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A REVIEW OF DIRECT DRIVE PROPORTIONAL ELECTROHYDRAULIC SPOOL VALVES: INDUSTRIAL STATE-OF-THE-ART AND RESEARCH ADVANCEMENTS

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ABSTRACT

This paper reviews the state of the art of directly driven proportional directional hydraulic spool valves, which are widely used hydraulic components in the industrial and transportation sectors. Firstly, the construction and performance of commercially available units are discussed, together with simple models of the main characteristics. The review of published research focusses on two key areas: investigations that analyse and optimize valves from a fluid dynamic point of view, and then studies on spool position control systems. Mathematical modelling is a very active area of research, including Computational Fluid Dynamics (CFD) for spool geometry optimisation, and dynamic spool actuation and motion modelling to inform controller design. Drawbacks and advantages of new designs and concepts are described in the paper.

1. Introduction

Proportional valves are critical components in many hydraulic actuation and power transmission systems. They are used where flowrate and hence actuation speed needs to be accurately controlled. Typical applications include mobile hydraulics (excavators, wheel loaders etc.), machine tools, industrial automation, and marine hydraulics. However the terms ‘proportional valve’, ‘servovalve’ and ‘direct-drive valve’ are not well defined and sometimes used interchangeably. In this paper, we are concerned with spool valves in which the spool is directly driven by an electrical actuator, specifically a proportional solenoid. Spool position both determines the direction of flow and modulates the flow rate. In contrast, a servovalve spool is driven by a faster, more powerful and more

linear actuator, typically a hydraulic pilot stage, and is manufactured to finer tolerances. Servovalves often have nominally zero overlap (dead band), whereas proportional valves are designed to have appreciable overlap, typically more than 3% [1].

A servovalve requires more precise manufacturing tolerances than a proportional valve, and to achieve this a servovalve is usually designed with the spool sliding in a bushing sleeve made of the same material as that of the spool. Detailed metering features in a servovalve can be obtained by providing the sleeve with slots. Instead, the spool in a proportional valve directly slides in the valve body, and notches and grooves are machined on the spool to achieve the desired flow rate trend versus spool position. The smaller overlap of servovalves is also synonymous with better response speed. This characteristic is further enhanced in servovalves by high speed spool actuation in which the pilot stage serves as a hydraulic amplification system capable of generating high pressure differences across the end faces of the main spool, which in consequence is moved by a very high actuation force. Instead, proportional directional valves are commonly moved directly by proportional solenoids, whose actuation forces are lower than those obtained in servovalves [1]. Note however that to control higher flowrates, multi-stage proportional valves may be used, which employ a small pilot spool to actuate a large main stage spool. In addition to the lower response speed (which is also due to the high moving masses of direct actuation systems), direct operated proportional directional valves are not capable of producing such a high “chip shear force”, namely, the force necessary to shear contamination particles that can be caught between the edges of a metering section. These drawbacks make proportional valves unsuitable for critical applications, such as aerospace (where the additional size and weight is also a problem). However, by virtue of their robustness and relatively low cost compared to servovalves, proportional valves are extensively used in many industrial applications. Unlike an on/off valve, a proportional valve can be instrumental in avoiding sudden acceleration and deceleration of an actuator, in addition to providing more accurate control of its position and/or velocity [1].

Given their importance in several industrial sectors, this paper discusses the state of the art of directly driven proportional directional valves. First, their operating principles and mathematical models used by researchers and industrial engineers to study and design these valves will be discussed. Then, an overview of commercially available valves will be given, with emphasis on their performance. Finally, a detailed review of the current research will be provided that is focused on the fluid dynamic analysis and on spool position control systems.

2. Operating principles and analytical modelling

Directly driven proportional directional valves have an inner sliding spool which is directly moved by either one solenoid or two solenoids placed at the spool extremities. The spool is provided with notches and grooves designed to achieve a desired flow rate trend as a function of the spool position [2]. These valves usually present a dead band given by the spool overlap, which can be as high as 10% or more of the spool stroke. In addition, usually the spool moves in a bore directly drilled in the valve body [1].

Fig. 1 shows a typical architecture of the most used valve typology, namely a 4 way 3 position (4/3) proportional valve along with its symbol. The sliding spool is pushed directly by either the right solenoid or the left solenoid depending on the required hydraulic connections (P-A and B-T or P-B and A-T) [3]. The oil enters the valve through the high pressure port P, then it flows through the metering section P-B or P-A (whose flow area is determined by both the metering notches in the spool and the opening degree) and finally exits the valve towards the actuator. Likewise, the oil discharged from the actuator re-enters the valve flowing through the metering section A-T₁ or B-T₂. Ports T₁ and T₂ are internally connected (not represented in Fig. 1 for simplicity) so as to form a single discharge port T [4].

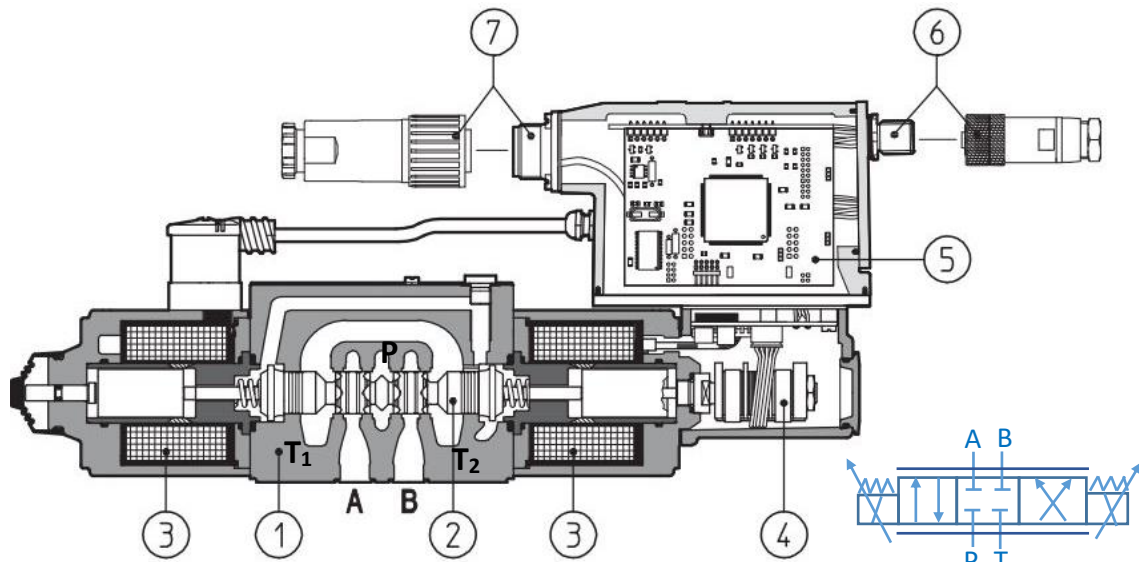


Fig.1. Proportional valve ATOS-DKZOR-T [5]

(1:valve body, 2:spool, 3:solenoid, 4:LVDT,5: electronic control, 6 and 7: connectors)

Analytical models were developed in the past [6] [7] and are currently used in scientific literature to easily study these valves [8] [9] [10] [11] [12]. The flow rate through a proportional directional valve depends on the opening area and on the pressure drop through the valve. If Δp is the pressure drop

measured across a metering edge and $A_r(x)$ the metering section area (a function of the spool position x), the volumetric flow rate Q through the metering chamber can be calculated as:

$$Q = C_d A_r(x) \sqrt{\frac{2\Delta p}{\rho}} \quad (1)$$

where C_d is the discharge coefficient of the metering section. The desired function $A_r(x)$ is obtained by properly designing the notches machined on the spool.

In the case of a 4/3 valve, a discharge coefficient must be defined for each of the two metering edges in the flow path. In such a case, the overall discharge coefficient through the valve can be calculated as proposed in [3]:

$$C_{d,V} = \frac{\sqrt{C_{d,1}^2 C_{d,2}^2}}{\sqrt{C_{d,1}^2 + C_{d,2}^2}} = \frac{Q}{A_r \sqrt{\frac{2\Delta p_v}{\rho}}} \quad (2)$$

where $C_{d,1}$ and $C_{d,2}$ are the discharge coefficients through the metering chambers and $C_{d,V}$ represents the overall flow coefficient of the valve, with Δp_v being the overall pressure drop through the valve. According to equations 1 and 2, it is evident that, for a given opening degree, the flow rate depends on the pressure drop through the valve. Fig. 2 shows qualitatively how the metering curve changes with the pressure drop through a valve. It refers to an overlapped valve (namely, a valve having the spool land longer than the adjacent gap in the valve body), which is the most common proportional valve typology [13], and the curve is determined by $A_r(x)$, given by the notch shape.

For a given opening degree, the change in the flow rate because of changes in the pressure drop can be calculated through equation (3), with subscripts 1 and 2 denoting two different operating conditions:

$$\frac{Q_1}{Q_2} = \sqrt{\frac{\Delta p_1}{\Delta p_2}} \quad (3)$$

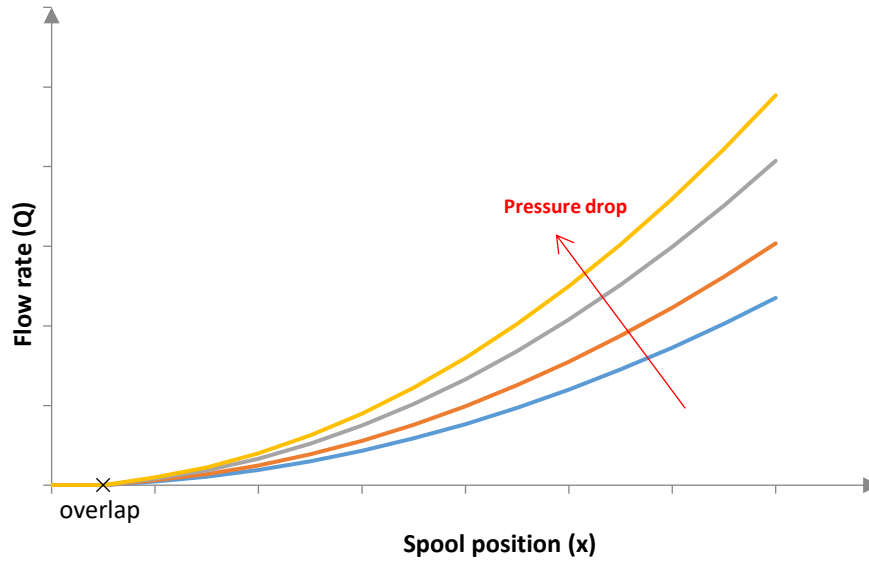


Fig.2. Metering curve as a function of the pressure drop for a given opening degree

Proportional valves can work in an open loop configuration or in a closed loop one, the latter employing a position sensor, typically a linear variable differential transformer (LVDT), for more precise control of the spool position [14]. Open loop control systems are cheaper, but are affected by changes in the operating conditions, as they rely on fixed parameters tuned for certain conditions.

In both cases, standard commercial electronic cards (shown in Fig.1) provide the solenoids with a Pulse width modulation (PWM) signal having a primary PWM signal frequency usually in the range 200–20000 Hz. A dither signal (square or sinusoidal wave with a frequency lower than the PWM frequency) is also used to keep the spool vibrating, thus overcoming stiction between the spool and the valve body bore [15].

The block diagram of a typical control system is shown in Fig. 3, adapted from [14]. As discussed in [14], a proportional–integral (PI) controller can be used for coil current control, which improves the static and dynamic characteristics of the valve. Similar to dither, the flutter signal generator has the purpose of reducing both friction and the magnetic hysteresis loop of the solenoid, improving the performance of the valve in demanding applications [14].

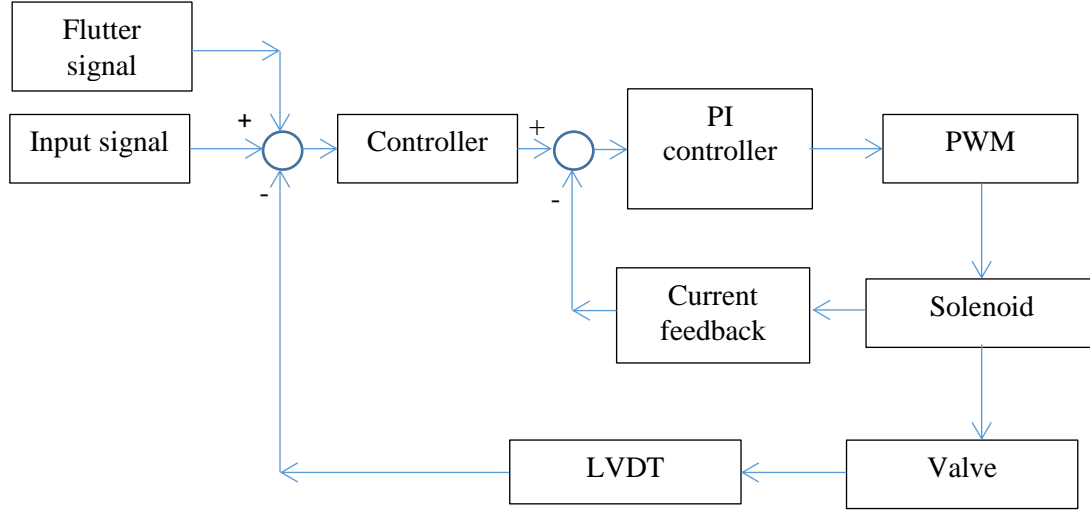


Fig.3. Typical control system of a proportional valve (adapted from [14])

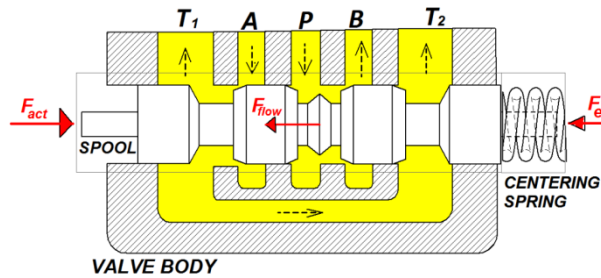
The current i flowing through the solenoids is therefore changed by varying the PWM duty cycle of voltage V applied to the solenoid coil, taking advantage of the resistive-inductive behaviour of the coil [16]:

$$V = iR + L \frac{di}{dt} \quad (4)$$

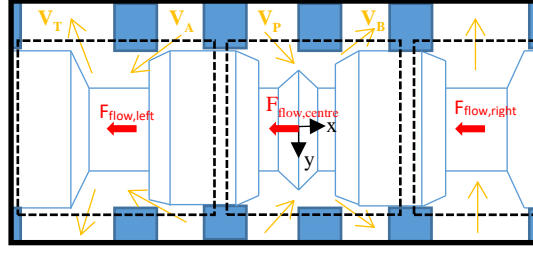
where R and L are the resistance and inductance of the coil. The higher the duty cycle of the PWM, the larger the average intensity of the current flowing through the solenoid and hence the higher the electromagnetic force exerted by the solenoid on the spool [17].

The electromagnetic force (F_{act}) acts in opposition to the damping force mainly due to the friction between the spool surface and the valve body surface ($F_{fr} = c\dot{x}$), the transient flow forces and stationary flow forces (F_{flow}) due to the fluid motion, and the elastic force ($F_{el} = k_{el}x$) produced by the centring springs (which are needed to maintain the spool in a centred position when no signal is applied to the coils). The resultant force accelerates the spool mass m :

$$F_{act} - F_{flow} - k_{el}x - c\dot{x} = m\ddot{x} \quad (5)$$



(a)



(b)

Fig.4. Section view of a 4/3 proportional valve (a) and enlargement on the spool surface with velocity and force vectors (b)

Fig. 4 provides a representation of a section view of a typical 4/3 valve with the spool being maintained in a fixed spool position x . The overall stationary flow force acting on the spool surface along the x axis is the sum of three contributions, due to the interaction between the fluid and the spool within the central chamber P-B ($F_{flow,centre}$), the left chamber A-T₁ ($F_{flow,left}$) and the right chamber in correspondence of the exit T₂ ($F_{flow,right}$). Each component is the sum of the pressure forces and viscous forces acting on the spool surfaces. As analyzed in [4], [6], [7], [17], [18], [19], the application of the conservation of momentum to the three control volumes shown in Fig. 4 leads to:

$$F_{flow} = F_{flow,left} + F_{flow,centre} + F_{flow,right} \cong \dot{m}[(V_A)_x - (V_T)_x] + \dot{m}[(V_B)_x - (V_P)_x] \quad (6)$$

where \dot{m} denotes the overall mass flow rate of the oil entering the valve; $(V_A)_x$ and $(V_T)_x$ are the average axial velocities at the inlet and outlet sections of the left control volume, respectively; $(V_B)_x$ and $(V_P)_x$ are the average axial velocities at the outlet and inlet of the central control volume, respectively. As the direction of the flow within the right control volume is orthogonal to the x axis, $F_{flow,right}$ can be neglected. In some models, a central conical surface (to be referred to as the compensation profile) and two lateral conical ones are constructed on the spool surface in order to increase the axial velocities $(V_T)_x$ and $(V_P)_x$, thus reducing the overall flow force acting on the spool [17].

In addition to the stationary flow forces, transient flow forces are developed during the spool movement from an initial position to a final one. As demonstrated in [20], the transient flow force in a metering chamber can be calculated as:

$$F_{flow,trans} = L \frac{d\dot{m}}{dt} \quad (7)$$

Where L is the axial distance between the inlet and outlet ports of the metering chamber.

Like all spool valves, proportional valves are vulnerable to a particular problem that is referred to as “hydraulic lock”, caused by an uneven pressure distribution around the circumference of the spool

which pushes the spool radially against the inner surface of its bore. Thus grooves are machined circumferentially around the spool to avoid an uneven pressure distribution and prevent hydraulic lock [21].

3. Commercially available proportional valves

Many manufacturers produce directly driven proportional directional valves, such as Atos [5], Parker [22], Bosch Rexroth [23], Moog [24], and Eaton [25]. Each model is usually provided by their manufacturer as a unique body which can be equipped with different sliding spools, according to the operation features required [2].

Commercially available proportional valves have less precise manufacturing tolerances than servovalves [8]. The larger tolerances on the spool geometry and spool overlap result in response nonlinearities, especially in the vicinity of neutral spool position [8].

A performance limitation is due to the direct actuation via proportional solenoids: these are relatively heavy and can generally operate in only one direction. Some solenoids are designed to operate in push-pull mode, but these are more expensive and generate lower driving forces than conventional ones [1]. In addition, for high pressures and/or flow rates required, the actuation force generated by commercially available solenoids is not high enough to counteract the opposing forces (flow forces + elastic forces of the centring springs). This results in a limited operational range for these valves as far as the maximum achievable flow rate is concerned [4]. The maximum flow rate is typically 100 l/min, with some models being capable of over 150 l/min, but only for low pressure drops. As an example, a large valve produced by ATOS [5] is the DKZOR-AES model, whose flow rate is about 105 l/min for a pressure drop of 30 bar through the valve. The solenoids employed in the DKZOR-AES model are very large and can have a maximum input power of 50 Watts. The operational field of the valve is reproduced in Fig.5: it is possible to observe that the maximum flow rate is about 160 l/min, but the pressure drop must be limited to 70 bar for such a flow rate level in order not to exceed the maximum power of the solenoids employed (higher pressure drop for the same flow would require a smaller metering flow area and so a higher flow velocity and a thus a higher flow force). The increase in the pressure drop causes a decrease in the maximum flow rate achievable; as shown by the blue curve of Fig.5, at 210 bar the maximum flow rate through the valve is lowered to about 90 l/min. This means that, for high pressure drops, it is not possible to reach the maximum opening degree of the valve, but only a part of the spool stroke can be used because of the limited power capability of the solenoids. A similar model, produced by ATOS, namely the DHZO-AES model [5], employs smaller solenoids with a lower maximum power produced, namely up to 30 Watts. The red curve in Fig. 5 reports the

operational field of the DHZO-AES; in spite of the similar geometric characteristics, the lower actuation power reduces the operation field of the valve, with a maximum flow rate of 64 l/min at 210 bar.

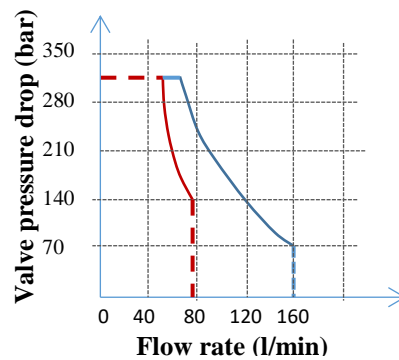


Fig 5. Operational field of two commercially available valves: DKZOR-AES (red line), DHZO-AES (blue line) [5].

With regard to the dynamic characteristics, they are only slightly affected by the operating pressure, unlike two stage valves. Available direct operated proportional directional valves have -90 deg phase lag frequency ranging from 10 Hz to 70 Hz ([5] [22] [23] [24] [25]). The higher values are obtained for small valves and those using closed loop controls. As an example, Fig. 6 shows a reproduction of the Bode plot of the proportional valve 4WREE size 10, produced by Bosch Rexroth [23]. This is a very large model, capable of achieving 150 l/min at 100 bar pressure drop. The Bode plot shows that the dynamic performance worsens when the amplitude of the input signal is increased, with the -90deg phase lag frequency varying from 20 Hz to 40 Hz for input signal amplitudes varying from 100% to 10% of the full stroke.

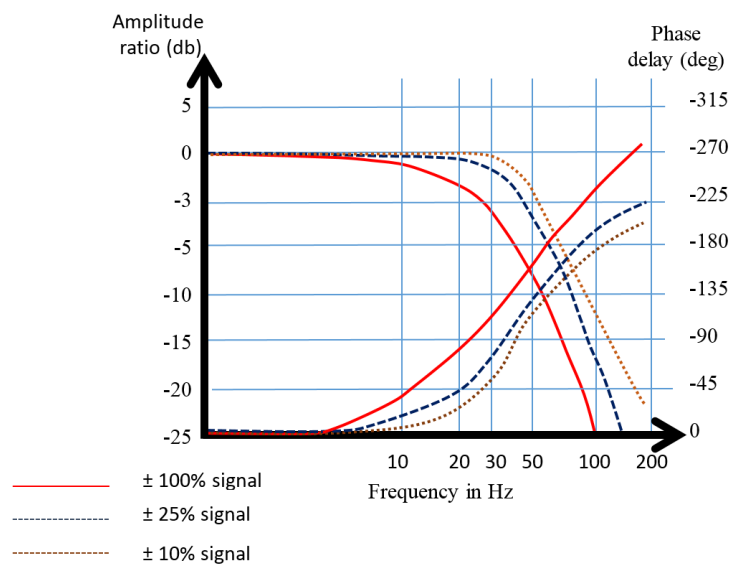


Fig 6. reproduction of the Bode plot of the proportional valve 4WREE size 10, produced by Bosch Rexroth [23]

Similarly, the response time to a step demand can vary from 10 ms to 50 ms according to the characteristics of the unit ([5] [22] [23] [24] [25]) and to the step amplitude. Fig. 7 shows the step tests for the valve 4WREE, achieved for 25%, 50%, 75% and 100% of the full stroke.

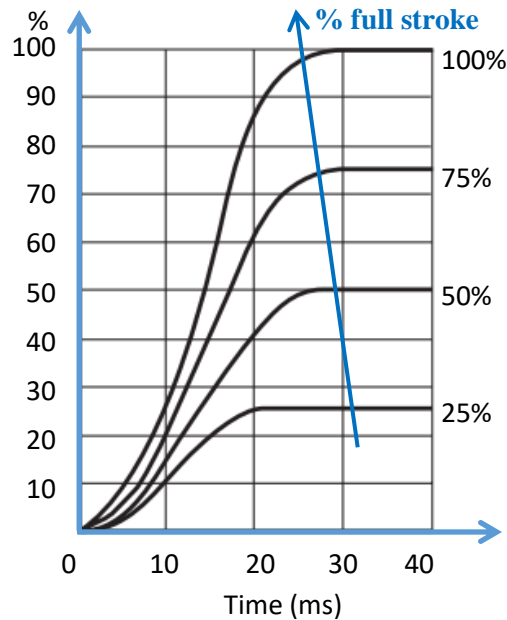


Fig 7. reproduction of the step test diagram of the proportional valve 4WREE size 10, produced by Bosch Rexroth [23]

Proportional directional direct operated valves are mainly used in the industrial and transportation sectors. They are not commonly used in aerospace, since in such application high response times and large actuation forces are required. The latter are necessary to avoid jammed spool conditions because of particle contamination (chip). A proportional solenoid is not capable of providing large actuation forces in order to shear a chip if it is jammed between the metering edges. Such a high force level can only be obtained through hydraulic amplification systems (e.g., nozzle flapper, Jet pipe, or deflector jet pilot stages) present in servovalves.

4. Fluid-dynamic research

It is common practise to consider, for preliminary calculations, a constant value of the discharge coefficient for proportional valves, with assumed values comprised between 0.65 and 0.7. However, as highlighted in [26], the discharge coefficient of a proportional valve is highly dependent on the notch geometry and spool position. In addition, the effects of cavitation are expected to affect the discharge coefficient, as discussed in [27].

Experimental and theoretical approaches have therefore been used to investigate the effects of the shapes of the notches upon the discharge coefficient and exit jet flow angle through a metering

section of a proportional valve. In [26], three notch shapes were experimentally analysed: the first one had a rectangular shape ended by a semicircle, the second one was obtained by connecting three semicircles with very short rectangles, while the third one had a triangular section (see Fig.8).

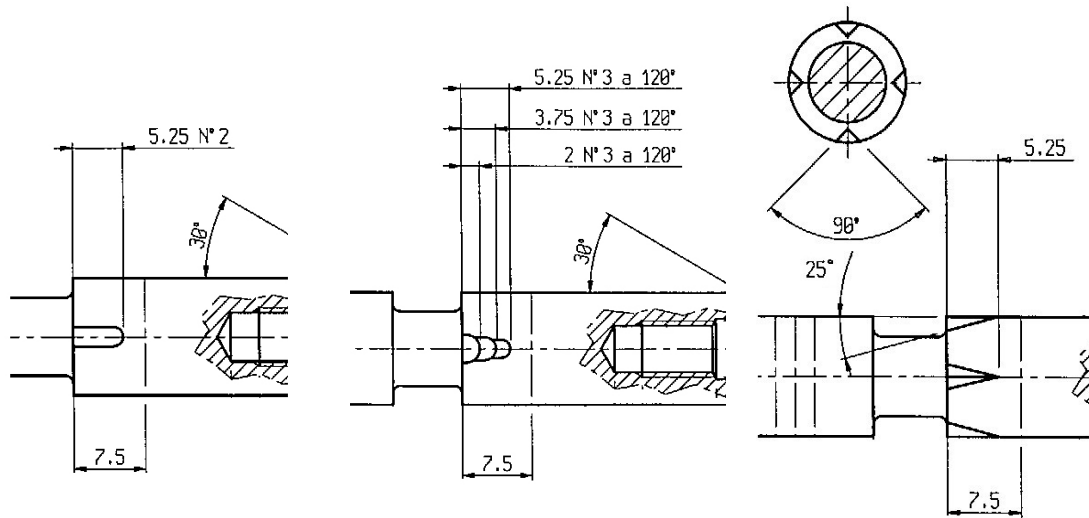


Fig 8. The three notch typologies analysed in [26]

The experimental circuit employed is reported in Fig. 9a, where a pump, a variable restrictor, a pressure relief valve, two flow meters and two pressure sensors were used to estimate the discharge coefficient. The experimental results of [26] showed that, for fully turbulent flow, the discharge coefficient assumed different values according to the notch type, number of notches employed and opening degree (spanning from 0.45 to 0.75). However, it was shown that, for a fixed opening degree, number and typology of notches, the discharge coefficient tends to a constant value with increasing Reynolds Number [26].

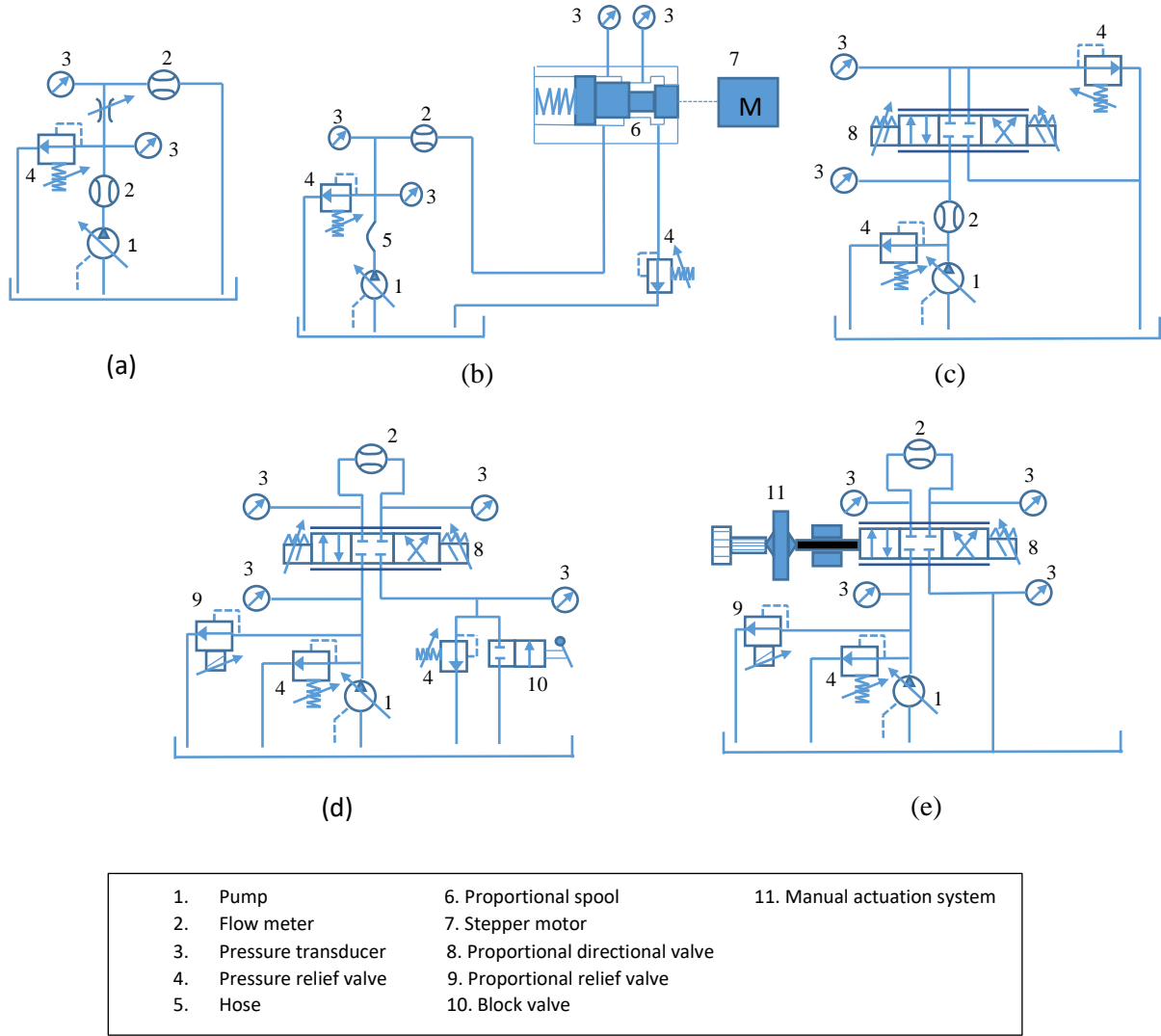


Fig 9. Experimental circuits employed in [26] : (a), [28]: (b), [29]: (c), [3]:(d), [17]: (e)

In addition to retrieving the discharge coefficients, experimental approaches have also been used to measure the flow forces. In [14] and [30], the flow forces were calculated as the difference between the solenoid force (F_{act}) and the force of the centring springs (F_{el}). The electromagnetic force in a proportional solenoid is a function of the armature position (coincident with the spool position x) and current i . In both cases, the force surface in the x, i plane was experimentally retrieved. In particular, the armature-solenoid-LVDT assembly was removed from the valve body and connected with a micrometer screw and a load cell in order to measure the actuation force as a function of the armature position and current, as shown in Fig. 10.

Figure 11 qualitatively shows the magnetic force surface as a function of the current and armature position. The maximum values of the actuation forces were measured to be around 100 N in [14] and

around 140 N in [30], evidencing that commercially available solenoids are not capable of developing high actuation forces compared to servovalves, whose actuation forces can be as high as 700 N [24].

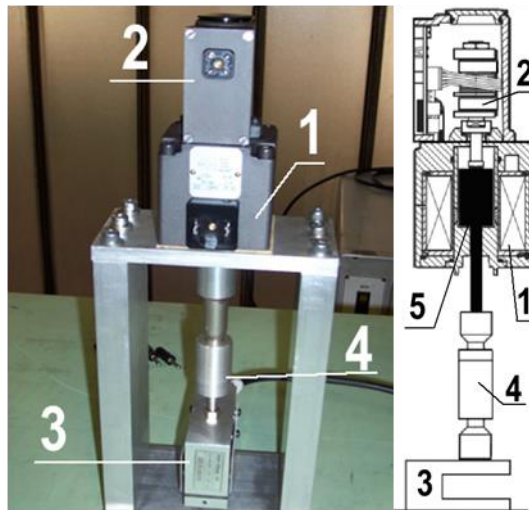


Fig 10. Experimental apparatus to evaluate the actuation forces: (1) coil , (2) LVDT , (3) load cell, (4) micrometer screw , (5) armature [30]

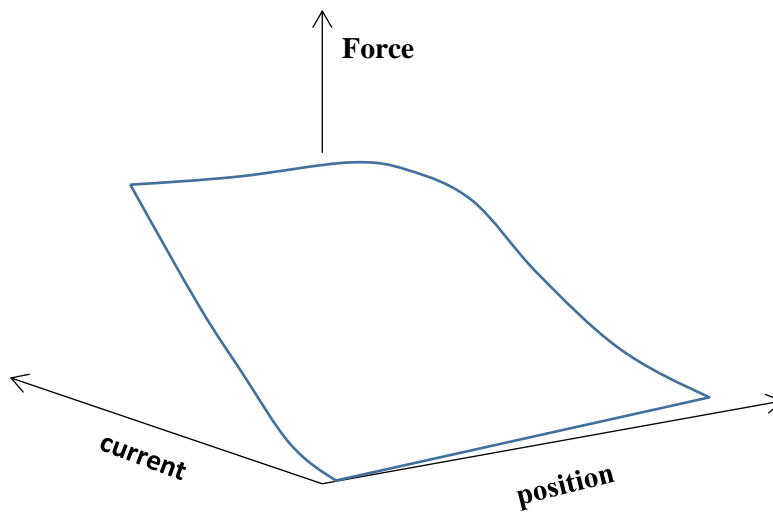


Fig 11. Electromagnetic force as a function of the armature position and current intensity

As an alternative approach, the actuation force was measured in [31] and [17] by using a manual actuation system: the armature inside the coil, which is in contact with the sliding spool, is moved through a knob; a load cell, interposed between the manual actuation and the armature, allows the actuation force to be measured (see Fig.12).

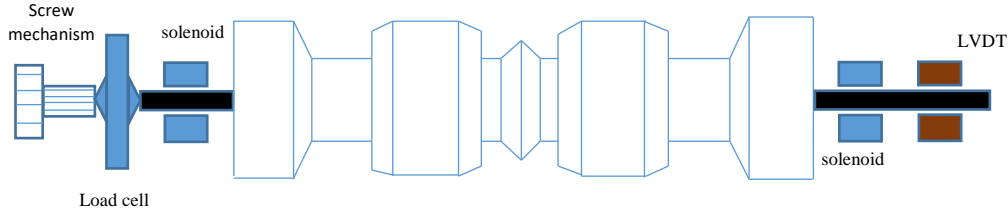


Fig 12. Experimental apparatus for measuring the actuation force that is based on a screw mechanism coupled with a force sensor

In addition to experimental approaches, a very effective method for analysis of flow through these valves is Computation Fluid Dynamics (CFD), available commercially as software tools such as Ansys Fluent [32]. The flow through a valve, supposed to be incompressible, can be modelled either by setting the values of pressure at the inlet and outlet or by setting the value of the velocity at the inlet [33]. The use of CFD modelling has proved its effectiveness for ON/OFF directional valves, with 2-Dimensional approaches and simplified computational domains being widely used to study the flow within these valves [34], [35], [36], [37], [38], [39], [40], [41]. However, the flow in a proportional valve is not axisymmetrical due to the presence of notches and grooves on the spool; for this reason, very detailed three dimensional approaches are commonly used to give more accurate results. Because of the domain complexity, unstructured grids are used for proportional valves. In [2], a partial three dimensional model (reproducing a circumferential sector of the entire valve), was used for different spool positions to study the flow field in a 4 way 3 position direct operated proportional directional valve. It was demonstrated in that paper that the use of small cylindrical notches on a spool with spherical notches can provide flow rate metering also at very small valve opening, while not influencing the overall flow forces acting on the spool. In addition, it was shown that the compensation profile (i.e. the central conical surface of the spool), if properly designed, can lead to a significant flow force reduction, with a negligible flow rate penalization at large openings. The flow force reduction is due to the increase in the axial component of the fluid velocity at the inlet section $(V_P)_x$, according to equation 6.

In [33], the high pressure chamber of a 4/3 proportional valve for load-sensing applications was simulated for 5 spool positions by using the open source code Open Foam [33]. The turbulence was modelled by means of the two zonal version of the $k-\omega$ model, known as the shear stress transport model. In particular, the effect of the direct and inverse flow through the notches of the metering chamber was investigated, showing that the discharge coefficient changes according to the fluid direction although a fixed geometry is considered. In addition, transient simulations were performed, in which the mesh motion was resolved by using a Generalized Grid Interface (GGI) approach [42],

originally developed for turbomachinery applications and modified to include not only rotational motion of the moving grid but also the linear displacement of a valve spool [33].

A partial three dimensional stationary model was also used in [28] to investigate the flow characteristics of three different groove profiles, namely the triangle shape, the U-shape and the spheroid shape, applied to a commercially available valve. The three groove profiles are shown in Fig. 13, where X is the spool opening; A_1 and A_2 are the axial and radial cross-sections respectively, and A_3 is the cross-section which crosses both the throttling edge and the lowest point of the groove. A_{\min} is the smallest cross-section across the throttling edge. Each groove has the length of the throttling grooves in the axial direction equal to 3 mm. The computational grid is shown in Fig. 14. An experimental circuit was assembled to validate the results, with the use of a stepper motor for a fine adjustment of the spool position (see Fig. 9b).

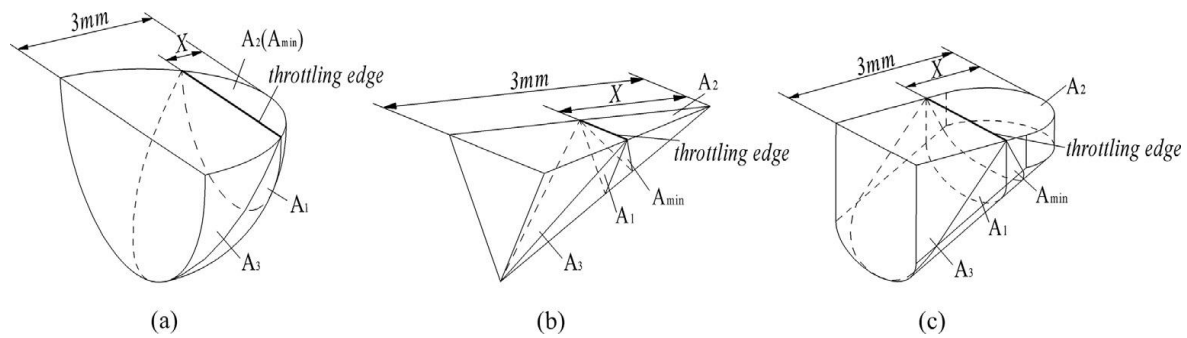


Fig.13. Geometric characteristics of: (a) the spheroid-shape groove; (b) the triangle-shape groove and (c) the divergent U-shape groove, analysed in [28]

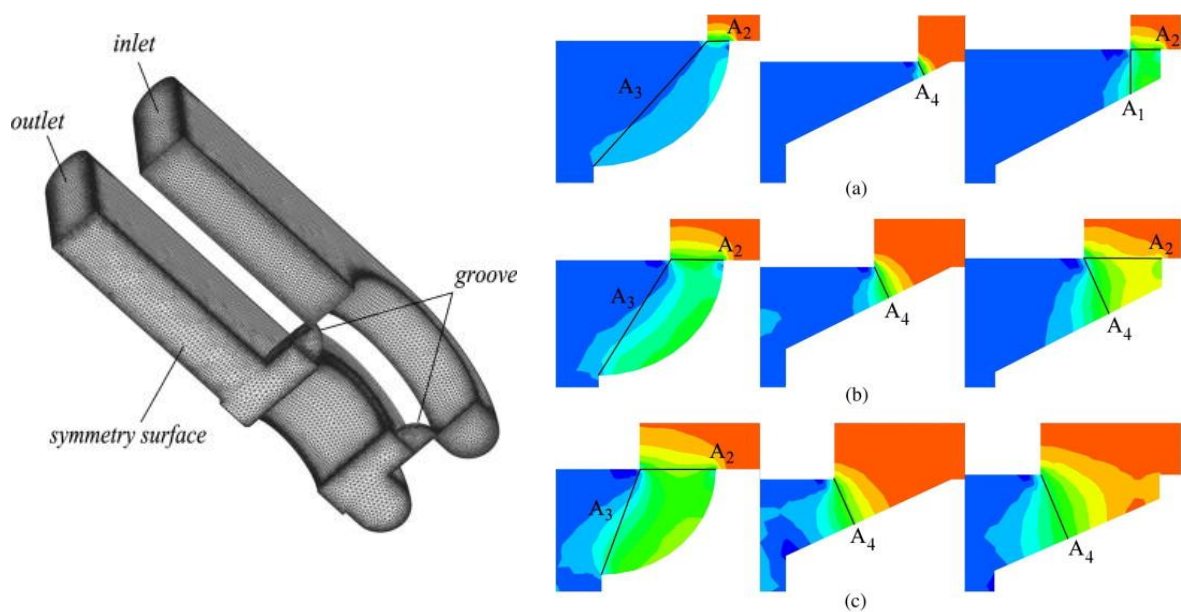


Fig.14. Left: Partial 3D CFD model employed in [28]; right: pressure contours on the symmetry plane: (a) $X = 0.6$ mm; (b) $X = 1.4$ mm; (c) $X = 2.0$ mm [28]

The results confirmed those obtained by [26], showing that the groove shape has significant effects on the discharge characteristics, the jet flow angle, the steady flow force and the throttling stiffness of the spool valve [28]. Fig. 14 also reports the pressure contours in the symmetrical surface of the notches at different spool positions. The pressure drop through the spheroid-shape groove is concentrated on cross-sections A_2 and A_3 within the entire range of the spool stroke, and with the increase of the opening, the proportion of the pressure drop changes gradually from cross-section A_2 to A_3 . For a triangular notch, the pressure drop is mainly centralized in cross-section A_4 [28]. The divergent U-shape groove has a more complex behaviour of the pressure distribution than the other two types, with the pressure drop distribution changing remarkably according to the spool position. The numerical and experimental results showed that the discharge coefficients first increase rapidly with the Reynolds number in the laminar flow region, and then gradually achieve the stable values around the transitional zones which range from 10 to 30 in terms of $Re^{0.5}$ [28]. This behaviour is qualitatively reproduced in Fig. 15. Table 1 reports the calculated asymptotic values of the discharge coefficients. Fig. 16 shows the values of the jet flow angles obtained via CFD for the different shapes of the grooves analysed. It is noteworthy that very different values are obtained and that the opening degree also significantly affects the discharge coefficients and flow angles [28]. This analysis is instrumental in pointing out the importance of CFD, which can allow a precise evaluation of the flow characteristics of a proportional valve for a given geometry of the spool notches.

| Opening (mm) | Discharge coefficient (asymptotic value) | | |
|-----------------|---|-----------------------------|--------------------------------|
| | Spheroid shape groove | Triangle shape groove | Divergent U-shape groove |
| 0.6 | 0.747 | 0.720 | 0.651 |
| 1.4 | 0.682 | 0.692 | 0.646 |
| 2.2 | 0.620 | 0.666 | 0.652 |

Table 1. Asymptotic value of the discharge coefficient according to different notch profiles (retrieved from [28])

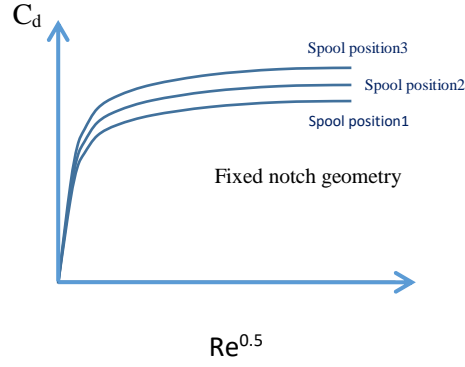


Fig.15. Qualitative trend of the discharge coefficient vs Reynold number for a fixed notch geometry

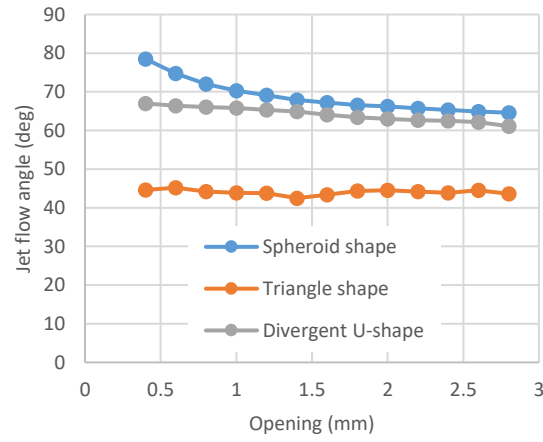


Fig.16. Values of the jet flow angle according to different notch profiles obtained via CFD[28]

A 3D CFD model reproducing a part of the spool surface was used to investigate the effects of other important geometrical features [21], such as the circumferential grooves machined on the spool surface. As only the zone in correspondence of the circumferential grooves was simulated, quadrilateral cells were generated. Circumferential grooves are fundamental to avoid “hydraulic lock” caused by an uneven pressure distribution on the spool surface during its movement. Hong and Kim suggested using spiral grooves instead of typical circumferential ones. Their work demonstrated that spool valves with spiral grooves could offer better performance in terms of relieving the asymmetric pressure distribution in the radial clearance because spiral grooves act as one continuous groove [21]. With the ever-increasing capability of computer hardware resources, the use of fully 3D approaches has become more common to study these valves via CFD [43]. Full 3D modelling was used in [29], [31], [44], [45], [46] to confirm at first that the use of constant values for the discharge coefficient may lead to significant errors, and then to obtain a method for calculating the coefficient values. Their numerical results were validated through the experimental circuit shown in Fig. 9c. In particular, functions for

evaluating the flow coefficient were proposed that depend on the spool position and on the flow rate. These functions can be particularly useful at the design stage in order to properly design the spool surface for a given metering curve.

The full 3D CFD analysis of [31] was carried out at the maximum opening of the WE10H valve produced by Bosch Rexroth to predict the stationary flow force, and this was calculated as a function of the flow rate. Fig. 17 shows the computational grid, whereas Table 2 reports the values of the predicted flow forces, extrapolated from [31]. The numerical results were also in very good agreement with experimental data, thus confirming the high accuracy reached by current full 3D methods.

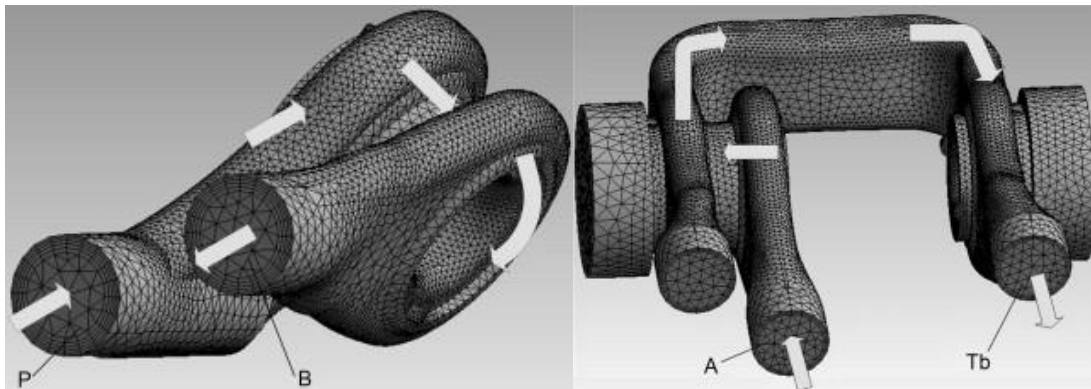


Fig.17. Full 3D grid developed in [31]

A further example of full 3D modelling is reported in [30]. In that work the aim was to increase the accuracy in the prediction of the stationary flow rate and flow forces compared to partially 3D models; 11 spool positions covering two-thirds of the full stroke were simulated for a commercially available valve, and 11 unstructured meshes with about 2 million cells were generated. The RNG k- ϵ model coupled with the enhanced wall treatment was implemented to resolve turbulence. The numerical predictions were compared with experimental results obtained through an experimental hydraulic circuit. The paper showed that a fully 3D discretization of the entire flow within the valve is required to properly predict the flow at small openings, where an axisymmetric approach fails, and in particular at large spool displacements, where also a partially 3D discretization had shown its limitations in previous papers.

| Flow rate (l/min) | Flow force (N) |
|-------------------|----------------|
| 30 | 10 |
| 60 | 20 |
| 90 | 58 |
| 120 | 80 |

| | |
|-----|-----|
| 150 | 122 |
|-----|-----|

Table 2. Flow force predicted at the maximum opening for a commercially available valve as a function of the flow rate (extrapolated from [31])

A full 3D model was also employed in [47] to evaluate the effects of small cylindrical notches to be machined on the spool surface of a commercially available valve manufactured by PONAR Wadowice [48]. Two spool versions with one notch and with two symmetrical notches were considered. Computational grids, each composed of about 4.1 million cells, were generated for gap widths spanning from 0.1mm to 0.4mm with a step of 0.1mm. The results confirm that the use of small cylindrical notches at the apex of main grooves can allow a proportional valve to operate with very low flow at small openings. In addition, it was demonstrated that the resultant axial force acting on the spool is significantly reduced because of the additional cylindrical notches.

A further improvement in the CFD modelling of proportional valves was given in [3], where the employed CFD model also accounted for cavitation, which is a non-negligible effect occurring in these valves. Among the available cavitation models, the Schnerr and Sauer model provided by Fluent was chosen since it is very robust and converges quickly. The results provided by the cavitation model were compared with the results obtained by the mono-phase model (in which the fluid was treated as incompressible). An experimental hydraulic circuit, shown in Fig. 9d, was also assembled in order to evaluate the effectiveness of the numerical model. A pressure relief valve was placed downstream of the proportional directional valve which allowed the pressure to be increased at port T, in order to evaluate the performance of the proportional valve without cavitation (due to the high discharge pressure). In contrast, cavitation could be generated by opening a block valve so that the hydraulic oil was able to by-pass the pressure relief valve, and the pressure at port T was decreased nearly to the atmospheric value [3]. Fig. 18 (a) shows the full 3D grid employed in the simulations. The “Porous Jump” boundary condition allowed a fixed pressure variation to be assigned through section C (see Fig. 18a), so as to simulate the pressure drop registered through the measuring equipment in the experimental tests. The pressure drop through the porous surface is computed by Fluent as

$$\Delta p_{porous} = - \left(\frac{\mu}{\delta} v + \frac{1}{2} c \rho v^2 \right) \Delta m \quad (8)$$

where δ is the permeability of the medium, c is the pressure-jump, v is the velocity normal to the porous face, ρ and μ are the density and the molecular viscosity of the fluid, respectively, and Δm is the thickness of the medium [3].

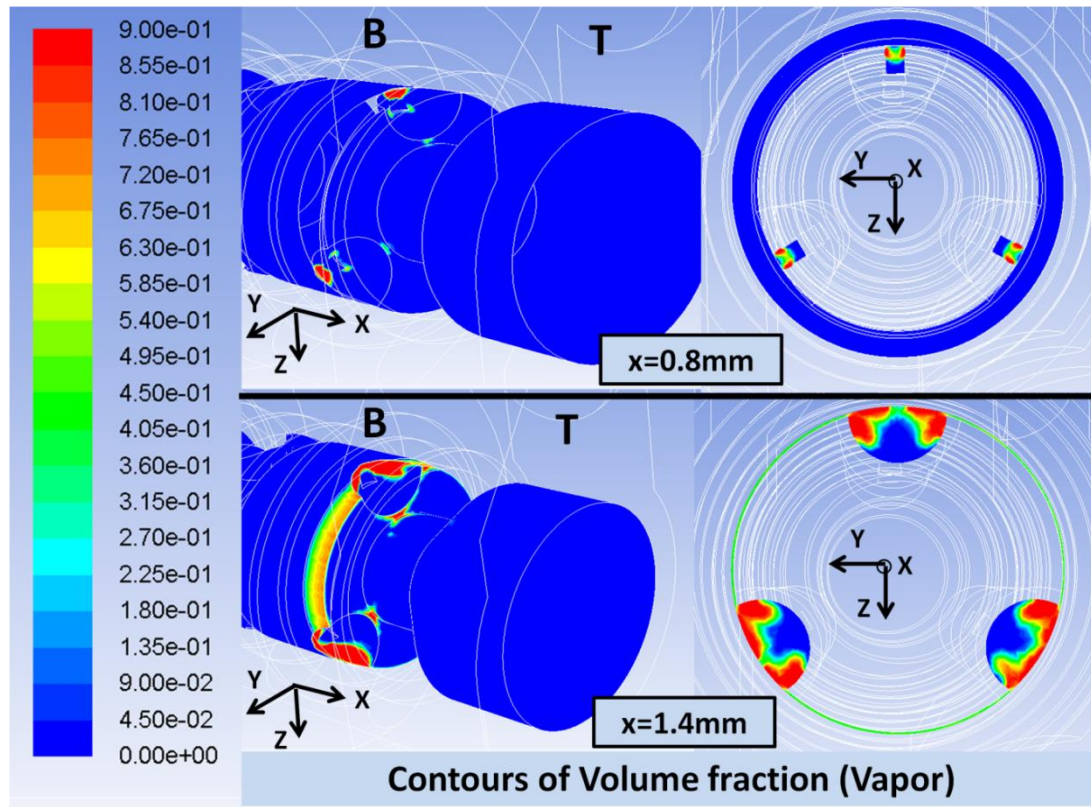
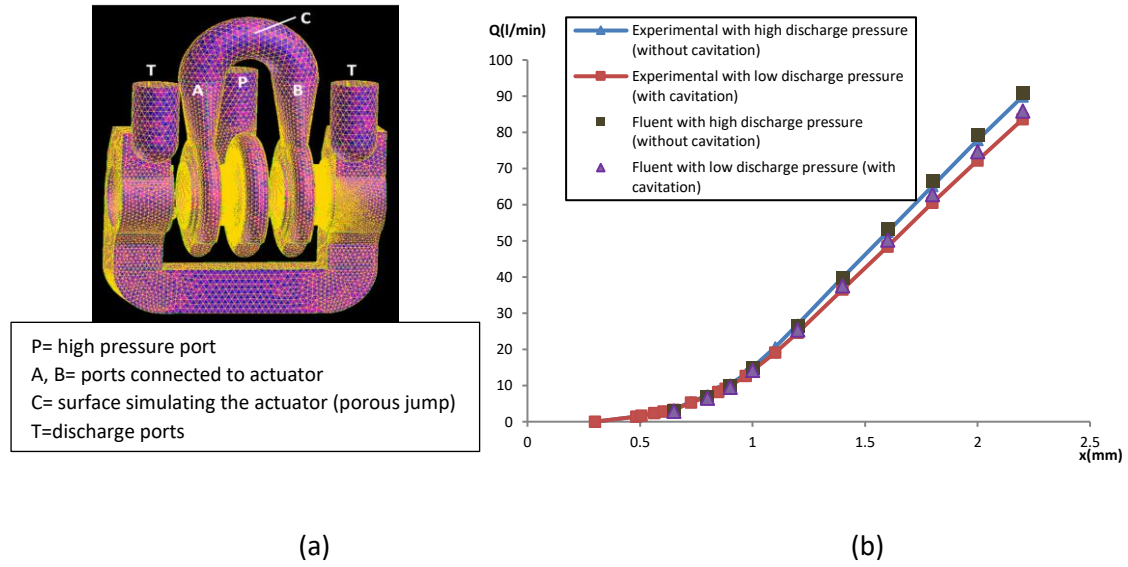


Fig.18. Full 3D grid used in [3] (a), metering curves obtained numerically and experimentally for low and high discharge pressure with an overall pressure drop=70 bar (b), and contours of volume fraction on the spool surface (c)

Fig. 18b shows the metering curves obtained experimentally and numerically for low discharge pressure and for high discharge pressure maintaining an overall pressure drop of 70 bar. It can be seen that cavitation affects the flow rate through the valve, causing a flow rate reduction of about 8% at

the maximum openings. The experimental results were very close to the numerical predictions, thus demonstrating that such a CFD model can reliably predict cavitation. Figure 18c provides the contours of the vapour volume fraction computed on the spool surface and on a section plane in correspondence of metering chamber B-T for the spool displacements equal to $x = 0.8$ mm and $x = 1.4$ mm, highlighting the importance of the phenomenon, especially at the large openings.

In [49], a full 3D CFD analysis was performed to study a new concept of valve. The solution presented in that paper uses an axial flow valve, where the oil passes through the valve along its axis, with two rotating surfaces causing a rotational metering. The result of that new design approach shows several advantages with respect to the common spool valves, such as the extremely compact size and the device versatility. This particular valve can realize the majority of the functions achievable using a two-way two-position proportional valve piloted by two pressure signals (for example a pressure compensated valve); the axial flow and the "built-in" metering edges yield the possibility to produce this valve as a cartridge component [49].

Such detailed 3D models can also be used at the design stage to obtain very effective geometries for the spool and for the valve body of standard valves in order to reduce the flow forces or cavitation intensity. In this regard, current research studies regarding spool valves are focused on reducing the flow force to extend their application range [4], [17], [18], [19], [50]. As pointed out in [50], commercially available valves present non-optimized geometries which restrict their potential, and opportune geometrical modifications to the valve body and spool are needed to minimize the stationary flow forces, which play a more important role in the control of higher hydraulic powers compared to the other resistant forces. In [18], effective changes were made both to the sliding spool and to the valve body of an ON/OFF small hydraulic seat valve, confirming that the non-optimized profiles of commercially available valves have a great influence on the required actuation forces. In [19], some possible methods were also provided to reduce the static flow forces in hydraulic sliding-spool on/off seat valves: the results of this research are very promising and prove that the axial component of the flow forces and therefore the necessary actuation force can be reduced significantly just by modifying the geometry of the valve housing and spool [4]. Although these results are concerned with ON/OFF valves, the same approach can be translated to proportional valves.

A genetic algorithm was coupled with a full 3D model of the flow field of a proportional valve in order to reduce the flow force at the maximum opening for a commercially available valve [4]. The geometrical parameters of the valve body were kept unchanged, whereas 4 geometrical parameters of the valve spool were selected as design parameters. The parameters, shown in Fig. 19, define the central and lateral surfaces of the spool, allowing the velocities at the inlet and outlet selections of the valve to be varied according to equation 6.

The comparison between the reference values and optimized ones is shown in Fig. 19. The optimized spool was constructed and experimentally compared with the reference one in [17]. A manual actuation system coupled with a load cell was used to measure the actuation force required by the two spools (see Fig. 9e and Fig. 12). An actuation force reduction of about 13% at the maximum opening was measured for a pressure drop of 70 bar through the valve.

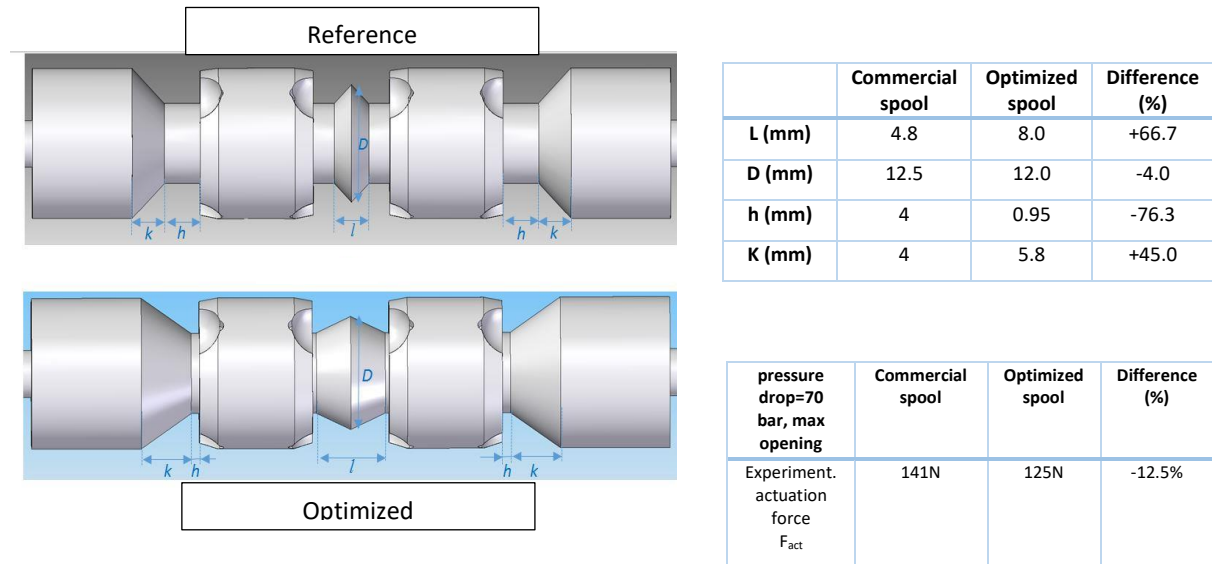


Fig.19. Design parameters adopted for the fluid dynamic optimization performed in [4] and experimentally validated in [17]: comparison between reference geometry and optimised one

| Authors | CFD software | Simulations | Domain | Number of cells | Fluid model | Turbulence model | Wall treatment |
|------------------------------|--------------|---|----------------------------|--|--|------------------|-------------------------|
| Amirante et al. 2007 [2] | Ansys Fluent | Full stroke, pressure drop=40 bar | Partially 3D, unstructured | 800 000 | Incompressible, stationary | RNG-k-ε | Enhanced wall treatment |
| Milani et al., 2012 [33] | Open Foam | 5 spool positions (inverse and direct flow) | Partially 3D, unstructured | 3 000 000 | Incompressible, stationary and transient | SST k-ω | Not mentioned |
| Amirante et al. 2014, a [30] | Ansys Fluent | 2/3 of the full stroke, pressure drop=100 bar | Fully 3D, unstructured | 2 000 000 | Incompressible, stationary | RNG-k-ε | Enhanced wall treatment |
| Lisowski et al., 2013 [31] | Ansys Fluent | Fixed position, simulation performed at 30, 60, 90, 120 and 150 l/min | Fully 3D, unstructured | 900 000 | Incompressible, stationary | K-ε | Not mentioned |
| Ye et al., 2014 [28] | Ansys Fluent | Five positions of the full stroke for different pressure drops | Partially 3D, unstructured | 450 000 | Incompressible, stationary | RNG-k-ε | Not mentioned |
| Amirante et al., 2014c [3] | Ansys Fluent | full stroke, pressure drop=70 bar | Fully 3D, unstructured | 1 500 000 to 2 000 000 according to the spool position | Mixture model, Schnerr and Sauer, stationary | RNG-k-ε | Enhanced wall treatment |
| Lisowski et al., 2018 [47] | Ansys Fluent | gap widths spanning from 0.1mm to | Fully 3D, unstructured | 4 100 000 cells | Incompressible, stationary | K-ε | Not mentioned |

| | | | | | | | |
|--|--|----------------------------|--|--|--|--|--|
| | | 0.4mm with a step of 0.1mm | | | | | |
|--|--|----------------------------|--|--|--|--|--|

Table 3. Some partially and fully 3D models presented in the scientific literature that simulate the flow field through a proportional directional valve

As an alternative approach for the flow force reduction, [31] suggested introducing additional channels in the valve body, without changing the spool geometry, as shown in Fig. 20. In that paper it was noted that the reduction of average velocity around the spool and better pressure compensation in the valve body are associated with lower flow force acting on the spool [31]. The numerical and experimental comparison between the reference valve and the novel one showed that in the latter the pressure is more balanced around the spool. A very large flow force reduction, up to 50%, was obtained with the additional channels.

The main settings of the 3D simulation studies analysed so far that predict the flow field through a proportional directional valve are summarized in Table 3.

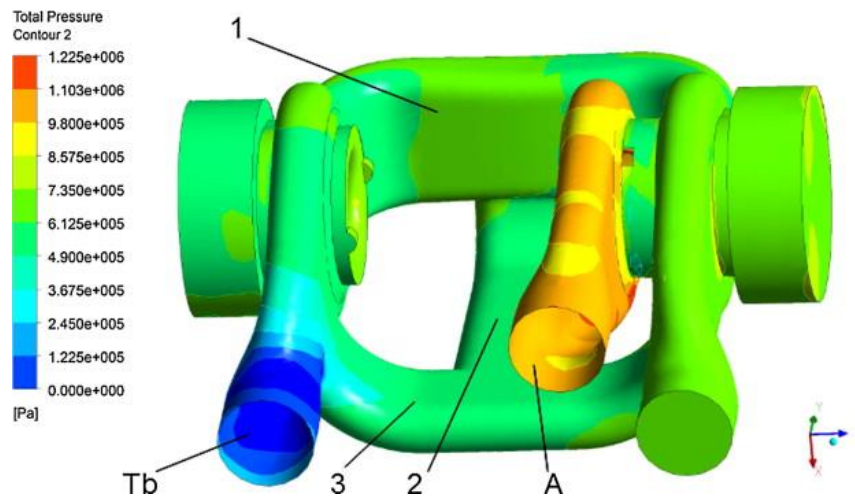


Fig. 20. Contours of pressure in the novel valve body geometry presented in [31]: 2 and 3 denote the additional channels

5. Research on control systems

The literature review analysed so far has been concerned with the fluid dynamic behaviour of proportional valves. In parallel to this, many studies have considered improving the spool position control systems for these valves, often with the help of models of the dynamic characteristics of spool actuation.

A nonlinear dynamic model was developed in [51], in which the solenoid was modelled as a nonlinear resistor/inductor combination, with inductance parameters changing according to the values of

displacement and current. Empirical curve fitting techniques were used to model the magnetic characteristics of the solenoid, enabling both current and magnetic flux to be simulated. The spool assembly was modelled as a spring/mass/damper system. The inertia and damping effects of the armature were incorporated in the spool model. The solenoid model was used to estimate the spool force in order to obtain a suitable damping coefficient value. The model accurately predicts both the dynamic and steady-state response of the valve to voltage inputs [51].

Analysis of a proportional solenoid was performed using finite element (FE) simulation in [52], by adopting an axially symmetrical two dimensional (2D) FE model in ANSYS/Emag. The electromagnetic force and flux linkage characteristics in all armature positions and for different currents were analysed. In [53], a discontinuous projection based adaptive robust controller was developed that is capable of compensating for the effect of the valve deadband, and certain straight-line approximations were used to model the nonlinear flow gain coefficient of the valve.

A novel nonlinear sliding mode controller was developed in [54]. The results demonstrated that the sliding mode controller can determine fast response times, with small overshoots and steady-state errors.

In Liu et al., 2011, a multi-domain nonlinear dynamic model of a proportional solenoid valve system was developed in the form of nonlinear state equations and was validated by experimental data. This model successfully predicts the dynamic characteristics of the valve and can be used as a powerful computational and simulation tool for valve design and algorithm optimization [55].

In [15], a control strategy that is based on the peak and hold (P&H) technique and that requires only a low cost microcontroller was proposed. The P&H technique, widely used to control Diesel fuel injection systems, consists of a particular PWM signal with a variable duty cycle composed of two constant signal phases with different voltage and current values. Similarly, the system proposed in that paper for the control of proportional valves employs, after the polarization phase, a peak high duty cycle to approach the target position of the spool followed by a lower duty cycle (hold) to maintain the position (see Fig. 21). This strategy gives a very fast valve response, even comparable to that provided by standard closed loop control systems, with a cost similar to available open loop control systems [15].

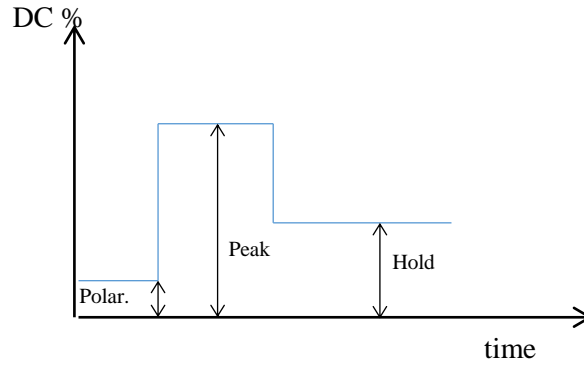


Fig. 21. Peak and hold technique employed in [15]

In [56], a digital state observer feedback control system, which is based on the digital signal processor and dynamic mathematical model of a proportional valve, was designed. [57] proposed three different types of controllers to improve the performance of proportional directional valves: a) an open loop compensator which requires the accurate valve dynamic model information; b) a full state feedback adaptive robust controller (ARC), which effectively takes into account the effect of parametric uncertainties and uncertain nonlinearities such as friction force and flow force; (c) an output feedback ARC controller to address the problem of unmeasurable states which takes into account the effect of both parametric uncertainties and uncertain nonlinearities [57].

A new method to tune the PID parameters of controllers for proportional directional valves without modelling and *a priori* knowledge of the system was proposed in [58]. The optimization of the PID parameters was achieved through statistical analysis by using the proposed method. The results showed that the optimized controller performs well, confirming that the tuning method is effective while also being straightforward to implement [58].

A combined Fuzzy-PID system was designed in [59] to control a proportional valve, showing fast response and small overshoot. In [14], a control method was proposed that uses the proportional solenoids simultaneously, contrary to the normal control method (NCM) that energizes only one solenoid at a time. The performance of the valve is greatly influenced by the nonlinearity of the proportional solenoid, such as dead zone and low force gain with a small current, and this effect cannot be eliminated by a simple dead-zone current compensation [14]. To avoid this disadvantage, the authors of that paper proposed a differential control method (DCM). By employing DCM, the controller outputs differential signals to simultaneously energize both solenoids of the proportional valve, and the operating point is found by analysing the force output of the two solenoids to minimize the variation of the current-force gain. The comparisons of the valve response characteristics were

performed between NCM and DCM by nonlinear dynamic simulation and experiments. Simulation and experimental results showed that, by using DCM, the frequency response of the valve is greatly enhanced, especially when the input is small, which means that the dynamic characteristics of the proportional valve are improved [14].

In [60], severe nonlinearities around the spool neutral position due to nonlinear spring force, nonlinear solenoid force and disturbances arising from flow force working on the spool were overcome through a dead-zone compensation design, a disturbance rejecting control design, and a control design for improving the reference tracking ability. In particular, an input shaping filter (ISF) was applied to optimise the control characteristics in the high frequency range [60]. The results demonstrated that the application of the proposed control design using ISF can satisfactorily compensate for the dead-zone, and greatly improve the dynamic response characteristics of the valve [60].

In [61], relationships of classical electrodynamics were used to derive a clear and detailed model that describes the influence of sinusoidal currents on the inductance and eddy current resistance, dependent on the spool position. In addition, a control theoretic approach was proposed to determine the spool position, and a numerically efficient reduced order observer was designed [61].

In [62], the application of a model-based control structure called Embedded Model Control (EMC) was presented. The overall control consists of two hierarchical loops: the inner loop is the solenoid current regulator with a closed loop bandwidth close to 1 kHz. The outer loop is a position tracking control, in charge of the accurate positioning of the spool with respect to valve openings [62].

In [63], a method for identifying solenoid valve transition events by analysing the current through the solenoid coils was proposed. The method estimates the spool position through identifying slope changes in the solenoid coil current traces. This methodology was able to identify the timing of valve transition events with less than 7% error compared to the measurement of the position of the valve spool obtained through a laser displacement sensor. As the method is based on measuring the current it requires no modification to the valve or valve housing [63].

In [64], an improved nonlinear sliding-mode controller was developed. Experimental studies were conducted and the results showed that the controller can achieve a continuous and stable sliding mode state, realize the time-optimal step response of a valve, and exhibit strong disturbance rejection abilities [64].

A new idea is represented by the “digital hydraulics” approach, which aims at applying digital principles to hydraulics by using multiple or high-speed on/off valves controlled through software rather than using proportional and servo-valves [65][66]. With regard to multiple valves, the idea is to use only one size of valve, which is optimal in the sense of flow density, response time and fault

tolerance. However, this concept is very demanding, as a larger number of valves is needed. The benefits of such systems are fast and amplitude independent response, redundancy, and robustness against oil impurities. Drawbacks are large space requirement, complexity of control and possibly higher price. Several papers are present in the scientific literature that study digital hydraulics systems in detail [67] [68] [69] [70] [71] [72] [73] [74] [44].

6. Conclusions

This paper has provided an overview on the operating principles, mathematical modelling, industrial and research state of the art of directly driven proportional directional hydraulic valves. These valves contain an inner sliding spool, directly moved by solenoids inside a valve body, which is provided with notches to allow flow rate metering as a function of the spool position. Commercially available units present lower dynamic performance but also lower costs than servovalves. The operational field on the flow rate-pressure plane is limited because sufficiently compact solenoids are not capable of counteracting high flow forces.

The flow rate and flow forces can be predicted by means of simple formulae which can be very useful for preliminary calculations. The validity of these formulae has been widely demonstrated; however, they depend on the discharge coefficient and flow angles, which can assume different values according to the spool position, notch geometry and operating conditions. To investigate how these parameters affect the discharge coefficient and flow angle, experimental and CFD approaches have been used in the scientific literature. The role of CFD modelling is noteworthy as it can allow a precise evaluation of the flow characteristics of a proportional valve for a given geometry of the spool notches, without the necessity of setting up an entire experimental circuit. At first, authors concentrated their efforts in developing partially 3D CFD models. However, with the ever-increasing capability of hardware resources, the use of fully 3D approaches has become more common, and the scientific literature presents several papers in which the high accuracy of full 3D models is demonstrated by comparing numerical predictions with experimental data. The flow is usually treated as incompressible, however some models also take into account the occurrence of cavitation by means of a mixture model which simulates the formation of vapour inside the liquid. The use of CFD to predict the discharge coefficient, flow angle and occurrence of cavitation can assist valve design by studying optimized geometry, in particular to allow the flow forces to be reduced, thus aiming at enlarging the operation field of these valves. A few optimized geometries are proposed in the literature that are more effective than commercial ones in terms of flow forces. These geometries regard either the optimization of the spool profile, by acting on the inlet and outlet velocity angles, or the valve body,

by adopting additional channels which can equalize the circumferential pressure distribution on the spool.

Research has also been focused on improving the control systems of these valves. Detailed models of the solenoid assembly are proposed to accomplish this task. Several effective methods have been proposed, such as a control based on the peak and hold technique, or a differential control method in which the controller outputs signals to simultaneously energize both solenoids of a proportional valve to improve linearity, or the employment of shaping filters or hierarchical control loops, or sliding mode controllers. A new idea is “digital hydraulics”, which aims at applying digital principles to hydraulics by using multiple or high-speed on/off valves controlled through software rather than using proportional or servo valves.

References

- [1] T. H. and N. Vaughan, “the Hydraulic Handbook,” vol. 3, p. 781, 1996.
- [2] R. Amirante, P. G. Moscatelli, and L. A. Catalano, “Evaluation of the flow forces on a direct (single stage) proportional valve by means of a computational fluid dynamic analysis,” *Energy Convers. Manag.*, vol. 48, no. 3, pp. 942–953, Mar. 2007.
- [3] R. Amirante, E. Distaso, and P. Tamburrano, “Experimental and numerical analysis of cavitation in hydraulic proportional directional valves,” *Energy Convers. Manag.*, vol. 87, 2014.
- [4] R. Amirante, L. A. Catalano, C. Poloni, and P. Tamburrano, “Fluid-dynamic design optimization of hydraulic proportional directional valves,” *Eng. Optim.*, vol. 46, no. 10, 2014.
- [5] “Atos SPA. <http://www.atos.com/>. Accessed September 2017.” .
- [6] J. F. Blackburn, G. Reethof, and J. L. Shearer, *Fluid power control, The Mit press and Wiley*. 1960.
- [7] H. . Merritt, *Hydraulic Control System*. 1967.
- [8] B. Eryilmaz and B. H. Wilson, “Unified modeling and analysis of a proportional valve,” *J. Franklin Inst.*, vol. 343, no. 1, pp. 48–68, 2006.
- [9] R. Amirante, A. Lippolis, and P. Tamburrano, “Theoretical and Experimental Analysis of a Coupled System Proportional Control Valve and Hydraulic Cylinder,” *Univers. J. Eng. Sci.*, vol. 1, no. 2, pp. 45–56, 2013.
- [10] W. Jiang, X. Huang, Q. Gao, and A. Peng, “The modeling and parameter identification of the hydraulic proportional valve,” in *2010 International Conference on Computer, Mechatronics, Control and Electronic Engineering, CMCE 2010*, 2010, vol. 3, pp. 482–485.
- [11] Y. Zhu and B. Jin, “Analysis and modeling of a proportional directional valve with nonlinear solenoid,” *J. Brazilian Soc. Mech. Sci. Eng.*, vol. 38, no. 2, pp. 507–514, 2016.
- [12] T. M. Menshawy, M. a Moghazy, and A. H. Lotfy, “Investigation of Dynamic Performance of an Electro-Hydraulic Proportional System,” *13th Int. Conf. Aerosp. Sci. Aviat. Technol. Int. Conf. Aerosp. Sci. Aviat. Technol.*, pp. 1–18, 2009.
- [13] A. Parr, *Hydraulics and Pneumatics (Third edition) A technician’s and engineer’s guide*.

- Butterworth-Heinemann, The Boulevard, Langford Lane, Kidlington, Oxford OX5 1GB, UK, 2011.
- [14] B. Jin, Y. Zhu, W. Li, D. Zhang, L. Zhang, and F. Chen, "A differential control method for the proportional directional valve," *J. Zhejiang Univ. Sci. C*, vol. 15, no. 10, pp. 892–902, 2014.
 - [15] R. Amirante, A. Innone, and L. A. Catalano, "Boosted PWM open loop control of hydraulic proportional valves," *Energy Convers. Manag.*, vol. 49, no. 8, pp. 2225–2236, 2008.
 - [16] K. F. Elmer and C. R. Gentle, "A parsimonious model for the proportional control valve," *Proc. Inst. Mech. Eng. Part C J. Mech. Eng. Sci.*, vol. 215, no. 11, pp. 1357–1363, 2001.
 - [17] R. Amirante, E. Distaso, and P. Tamburrano, "Sliding spool design for reducing the actuation forces in direct operated proportional directional valves: Experimental validation," *Energy Convers. Manag.*, vol. 119, 2016.
 - [18] M. Simic and N. Herakovic, "Reduction of the flow forces in a small hydraulic seat valve as alternative approach to improve the valve characteristics," *Energy Convers. Manag.*, vol. 89, pp. 708–718, 2015.
 - [19] N. Heraković, "CFD simulation of flow force reduction in hydraulic valves," *Teh. Vjesn. - Tech. Gaz.*, vol. 22, no. 2, pp. 453–463, 2015.
 - [20] Q. Yuan and P. Y. Li, "An experimental study on the use of unstable electrohydraulic valves for control," in *Proceedings of the American Control Conference*, 2002, vol. 6, pp. 4843–4848.
 - [21] S. H. Hong and K. W. Kim, "A new type groove for hydraulic spool valve," *Tribol. Int.*, vol. 103, pp. 629–640, 2016.
 - [22] "PARKER HANNIFIN CORP. <http://ph.parker.com/us/en/proportional-valves>. Accessed September 2017."
 - [23] "Bosch Rexroth USA. <https://www.boschrexroth.com/en/us/products/product-groups/industrial-hydraulics/proportional-and-servo-valves/index>. Accessed September 2017."
 - [24] "Moog controls ltd. <http://www.moog.com/products/servovalues-servo-proportional-valves.html>. Accessed September 2017."
 - [25] "Eaton. <http://www.eaton.com>. Accessed September 2017."
 - [26] M. Borghi, M. Milani, and R. Paoluzzi, "Influence of notch shape and number of notches on the metering characteristics of hydraulic spool valves," *Int. J. Fluid Power*, vol. 6, no. 2, pp. 5–18, 2005.
 - [27] J. R. Valdés, J. M. Rodríguez, R. Monge, J. C. Peña, and T. Pütz, "Numerical simulation and experimental validation of the cavitating flow through a ball check valve," *Energy Convers. Manag.*, vol. 78, pp. 776–786, 2014.
 - [28] Y. Ye, C. B. Yin, X. D. Li, W. J. Zhou, and F. F. Yuan, "Effects of groove shape of notch on the flow characteristics of spool valve," *Energy Convers. Manag.*, vol. 86, pp. 1091–1101, 2014.
 - [29] E. Lisowski and G. Filo, "Analysis of a proportional control valve flow coefficient with the usage of a CFD method," *Flow Meas. Instrum.*, vol. 53, no. December 2016, pp. 269–278, 2017.
 - [30] R. Amirante, L. Andrea Catalano, and P. Tamburrano, "The importance of a full 3D fluid dynamic analysis to evaluate the flow forces in a hydraulic directional proportional valve,"

- Eng. Comput.*, vol. 31, no. 5, pp. 898–922, 2014.
- [31] E. Lisowski, W. Czyzycki, and J. Rajda, “Three dimensional CFD analysis and experimental test of flow force acting on the spool of solenoid operated directional control valve,” *Energy Convers. Manag.*, vol. 70, pp. 220–229, 2013.
 - [32] Ansys, “Ansys fluent 12.0,” *Inc Northbrook*, no. April, pp. 49–53, 2009.
 - [33] M. Milani, L. Montorsi, and F. Paltrinieri, “Multidimensional Design of Hydraulic Components and Systems,” *Appl. Comput. Fluid Dyn.*, 2012.
 - [34] M. Borghi, M. Milani, and R. Paoluzzi, “Stationary axial flow force analysis on compensated spool valves,” *Int. J. Fluid Power*, vol. 1, no. 1, pp. 17–25, 2000.
 - [35] R. Amirante, G. Del Vescovo, and A. Lippolis, “Flow forces analysis of an open center hydraulic directional control valve sliding spool,” *Energy Convers. Manag.*, vol. 47, no. 1, pp. 114–131, 2006.
 - [36] R. Amirante, G. Del Vescovo, and A. Lippolis, “Evaluation of the flow forces on an open centre directional control valve by means of a computational fluid dynamic analysis,” *Energy Convers. Manag.*, vol. 47, no. 13–14, pp. 1748–1760, 2006.
 - [37] X. Baudry and J. C. Mare, “Linking CFD and lumped parameters analysis for the design of flow compensated spool valves,” in *Proceedings of the 1st FPNI-PhD symposium Hamburg*, 2000, pp. 249–58.
 - [38] G. Del Vescovo and A. Lippolis, “CFD analysis of flow forces on spool valves,” in *Proceedings of the 1st international conference on computational methods in fluid power technology*, 2003.
 - [39] G. Del Vescovo and A. Lippolis, “Three-dimensional analysis of flow forces on directional control valves,” *Int. J. Fluid Power*, vol. 4, no. 2, pp. 15–24, 2003.
 - [40] J. R. Valdés, M. J. Miana, J. L. Núñez, and T. Pütz, “Reduced order model for estimation of fluid flow and flow forces in hydraulic proportional valves,” *Energy Convers. Manag.*, vol. 49, no. 6, pp. 1517–1529, 2008.
 - [41] A. Posa, P. Oresta, and A. Lippolis, “Analysis of a directional hydraulic valve by a Direct Numerical Simulation using an immersed-boundary method,” *Energy Convers. Manag.*, vol. 65, pp. 497–506, 2013.
 - [42] M. Beaudoin and H. Jasak, “Development of a Generalized Grid Interface for Turbomachinery simulations with OpenFOAM,” in *Open Source CFD International Conference*, 2008.
 - [43] E. Frosina, A. Senatore, D. Buonoa, M. Pavanetto, and M. Olivetti, “3D CFD transient analysis of the forces acting on the spool of a directional valve,” *Energy Procedia*, vol. 81, pp. 1090–1101, 2015.
 - [44] E. Lisowski and J. Rajda, “CFD analysis of pressure loss during flow by hydraulic directional control valve constructed from logic valves,” *Energy Convers. Manag.*, vol. 65, pp. 285–291, 2013.
 - [45] E. Lisowski, G. Filo, and J. Rajda, “Pressure compensation using flow forces in a multi-section proportional directional control valve,” *Energy Convers. Manag.*, vol. 103, pp. 1052–1064, 2015.
 - [46] E. Lisowski and G. Filo, “CFD analysis of the characteristics of a proportional flow control valve with an innovative opening shape,” *Energy Convers. Manag.*, vol. 123, pp. 15–28, 2016.

- [47] E. Lisowski, G. Filo, and J. Rajda, "Analysis of Flow forces in the initial phase of throttle gap opening in a proportional control valve," *Flow Meas. Instrum.*, vol. 59, no. December 2017, pp. 157–167, 2018.
- [48] "PONAR Wadowice. <http://www.ponar-wadowice.com/directional-control-valves>. Accessed September 2017."
- [49] G. ANSALONI, P. MARANI, and R. PAOLUZZI, "Proportional Valve With Axial Flow and Rotational Metering," *Proc. JFPS Int. Symp. Fluid Power*, vol. 2008, no. 7–3, pp. 591–596, 2008.
- [50] N. Heraković, "Flow-Force Analysis in a Hydraulic Sliding-Spool Valve," *Strojarstvo*, vol. 51, no. 6, pp. 555–564, 2009.
- [51] N. D. Vaughan and J. B. Gamble, "The modeling and simulation of a proportional solenoid valve," *Trans. Soc. Mech. Eng. J. Dyn. Syst. Meas. Control*, vol. 118, pp. 120–125, 1996.
- [52] F. Bayat, A. Fadaie Tehrani, and M. Danesh, "Static and dynamic simulation of a proportional control valve using ANSYS /Emag," in *2011 IEEE International Conference on Mechatronics, ICM 2011 - Proceedings*, 2011, pp. 581–584.
- [53] F. Bu and B. Yao, "Nonlinear adaptive robust control of hydraulic actuators regulated by proportional directional control valves with deadband and nonlinear flow gains," in *Proceedings of the American Control Conference*, 2000, vol. 6, pp. 4129–4133.
- [54] J. B. Gamble and N. D. Vaughan, "Comparison of sliding mode control with state feedback and PID control applied to a proportional solenoid valve," *J. Dyn. Syst. Meas. Control*, vol. 118, no. 3, pp. 434–438, 1996.
- [55] Y. F. Liu, Z. K. Dai, X. Y. Xu, and L. Tian, "Multi-domain modeling and simulation of proportional solenoid valve," *J. Cent. South Univ. Technol. (English Ed.)*, vol. 18, no. 5, pp. 1589–1594, 2011.
- [56] X. Kong and H. Wang, "A spool displacement control system of proportional valve based on digital observer," *Key Eng. Mater.*, vol. 455, pp. 110–115, 2010.
- [57] F. Bu and B. Yao, "Performance Improvement of Proportional Directional Control Valves: Methods and Experiments," *Asme*, vol. xx, pp. 1–8, 2000.
- [58] B. Jin, Y. G. Zhu, and W. Li, "PID parameters tuning of proportional directional valve based on multiple orthogonal experiments method: method and experiments," *Appl. Mech. Mater.*, vol. 325, pp. 1166–1169, 2013.
- [59] X. Wang, "Fuzzy-PID control in electro-hydraulic proportional valve system," in *2010 IEEE 11th International Conference on Computer-Aided Industrial Design and Conceptual Design, CAID and CD'2010*, 2010, vol. 2, pp. 1454–1457.
- [60] I. Y. Lee, D. H. Oh, S. W. Ji, and S. N. Yun, "Control of an overlap-Type proportional directional control valve using input shaping filter," *Mechatronics*, vol. 29, pp. 87–95, 2015.
- [61] T. Braun, J. Reuter, and J. Rudolph, "Position Observation for Proportional Solenoid Valves by Signal Injection," *IFAC-PapersOnLine*, vol. 49, no. 21, pp. 74–79, 2016.
- [62] W. Acuña-Bravo, E. Canuto, M. Agostani, and M. Bonadei, "Proportional electro-hydraulic valves: An Embedded Model Control solution," *Control Eng. Pract.*, vol. 62, pp. 22–35, 2017.
- [63] A. C. Yudell and J. D. Van De Ven, "Predicting solenoid valve spool displacement through current analysis," *Int. J. Fluid Power*, vol. 16, no. 3, pp. 133–140, 2015.

- [64] J. hui Fang, F. Guo, Z. Chen, and J. hua Wei, "Improved sliding-mode control for servo-solenoid valve with novel switching surface under acceleration and jerk constraints," *Mechatronics*, vol. 43, pp. 66–75, 2017.
- [65] N. P. Sell, D. N. Johnston, A. R. Plummer, and S. Kudzma, "Control of a fast switching valve for digital hydraulics," *13th Scand. Int. Conf. Fluid Power*, pp. 497–503, 2013.
- [66] N. P. Sell, N. Johnston, A. R. Plummer, and S. Kudzma, "Development of a position controlled digital hydraulic valve," in *Proceedings of the ASME/BATH 2015 Symposium on Fluid Power and Motion Control FPMC2015 October 12-14, 2015, Chicago, Illinois, USA*.
- [67] S. S. Raghuraman, D. R. S. Kumar, "Digital hydraulic valves-a brief review," *J. Chem. Pharm. Sci.*, vol. 974, 2016.
- [68] M. Linjama, M. Paloniitty, L. Tiainen, and K. Huhtala, "Mechatronic Design of Digital Hydraulic Micro Valve Package," *Procedia Eng.*, vol. 106, pp. 97–107, 2015.
- [69] M. Paloniitty, M. Linjama, and K. Huhtala, "Equal Coded Digital Hydraulic Valve System - Improving Tracking Control with Pulse Frequency Modulation," *Procedia Eng.*, vol. 106, pp. 83–91, 2015.
- [70] M. Linjama and K. Huhtala, "Digital hydraulic power management system – towards lossless hydraulics," *Third Work. Digit. Fluid Power, Oct. 13 - 14, 2010, Tampere, Finl.*, pp. 5–22, 2010.
- [71] M. Linjama, A. Laamanen, and M. Vilenius, "On the pressure peak minimization in digital hydraulics," *Int. J. Fluid Power*, vol. 11, no. 1, pp. 2–16, 2007.
- [72] M. Huova, M. Karvonen, V. Ahola, M. Linjama, and M. Vilenius, "Energy Efficient Control of Multiactuator Digital Hydraulic Mobile Machine," *7th Int. Fluid Power Conf.*, pp. 1–12, 2010.
- [73] M. Linjama, "Digital Fluid Power – State of the Art," *12th Scand. Int. Conf. Fluid Power, May 18-20, 2011, Tampere, Finl.*, p. 331–xx, 2011.
- [74] M. Linjama, A. Laamanen, and M. Vilenius, "Is it Time for Digital Hydraulics?," *Proc. 8th Scand. Int. Conf. Fluid Power, SICFP '03, Vol1*, pp. 347–366, 2003.